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ENGINE TESTS OF PRESSURIZED SHUNT-TYPE COOLING SYSTEMS FOR A LIQUID-COOLED ENGINE

By Eugene J. Manganiello, Bruce T. Lundin
and John H. Povolny

Aircraft Engine Research Laboratory
Cleveland, Ohio

NACA

WASHINGTON

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SUMMARY

Tests have been conducted to determine the performance of two pressurized shunt-type cooling systems for liquid-cooled aircraft engines using a mixture of 30 percent AN-E-2 ethylene glycol and 70 percent water as the coolant. One of the systems (system A) employed an expansion tank typical of those in current use on unpressurized AN-E-2 ethylene glycol systems; the other system (system B) used an expansion tank designed by the Linde Air Products Company and modified for production by the Bell Aircraft Corporation. Neither system incorporated a venturi nor other pressure-boost arrangement at the pump inlet. Coolant-flow rates were determined for both systems over (a) a range of engine speeds at constant pump-inlet pressure and (b) a range of expansion-tank pressures at constant engine speed.

The results of tests of the two cooling systems show that with the expansion tank of system B higher pump-inlet pressures (approximately 6 lb/sq in.) and attendant better pump performance may be obtained than with the expansion tank used in system A for any fixed expansion-tank pressure. For an expansion-tank pressure equal to the vapor pressure of the coolant at a block-outlet temperature of 250° F, a coolant-flow rate of 255 gallons per minute is obtained with system B as compared with a flow rate of 200 gallons per minute obtained with system A. Both of these flow rates are adequate to cool 12-cylinder liquid-cooled engines at present power ratings. The higher pump-inlet pressures of system B, however, resulted in a more stable and safer operation of the coolant pump than could be obtained with system A.

INTRODUCTION

Recent increases in the power output of liquid-cooled aircraft engines have indicated the necessity for improvement in the cooling

of this type of engine. Although satisfactory cooling of American-built engines has been obtained with an unpressurized shunt-type cooling system using AN-E-2 ethylene glycol as the coolant, the results of unpublished tests conducted at the NACA Cleveland laboratory on an Allison V-1710 engine indicate that the cylinder-head temperatures in this engine approach limiting values for safe operation at present power ratings and exceed them at projected higher power ratings. The results of these tests also indicate that appreciable reductions in cylinder-head temperatures may be obtained by the use of a mixture of 30 percent ethylene glycol and 70 percent water as the coolant.

The relatively low boiling point of the 30-70 glycol-water mixture necessitates the use of a pressurized cooling system if coolant temperatures (250° F and higher) compatible with reasonable radiator size are to be used. Pressurized systems of the series type have been utilized for some time in British Rolls Royce engine installations and more recently in the American-built Packard versions of the Rolls Royce engine. Pressurized systems of the shunt type theoretically possess inherent advantages over pressurized systems of the series type; the principal advantages of the shunt-type system are higher pump-inlet pressures and concomitant superior pump performance for a given expansion-tank pressure. No experimental data are available, however, on their performance.

As part of a research program being conducted by the NACA on the cooling of liquid-cooled engines, the performance of pressurized shunt-type cooling systems has been investigated by means of engine tests using a mixture of 30 percent AN-E-2 ethylene glycol and 70 percent water as the coolant. Two different shunt-type systems were tested: system A used an expansion tank typical of those in current use on unpressurized AN-E-2 ethylene glycol systems; system B used an expansion tank designed by the Linde Air Products Company and modified for production by the Bell Aircraft Corporation. The tests described herein were conducted at Cleveland during June and July 1944 and consisted of a determination of coolant-flow rates over (a) a range of engine speeds at constant pump-inlet pressure and (b) a range of expansion-tank pressures at constant engine speed.

THEORY OF COOLING SYSTEMS

Much of the theory on cooling systems discussed herein has been presented in references 1 and 2. The satisfactory operation of a cooling system for a liquid-cooled aircraft engine depends upon the ability of the system to keep the flow rate as high as required to maintain the engine temperatures within safe limits. In order to

obtain adequate coolant-flow rates and stable pump operation it is necessary that the pressure at the pump inlet be higher than the vapor pressure of the coolant by an amount sufficient to prevent excessive pump cavitation. The tendency of a pump to cavitate is usually considered as a function of the "pressure proximity" at the pump inlet; the pressure proximity is defined as the difference between the absolute static pressure of the liquid and the vapor pressure of the liquid at the same point. This parameter may be used to correlate the effect of liquid composition and temperature on the cavitation characteristics of a pump.

Because the decrease of flow rate caused by pump cavitation is due to the formation of vapor within the pump, the presence of entrained air or vapor in the coolant will have the same effect on the pump performance as too low an inlet pressure. It is therefore essential that satisfactory air-vapor separators be incorporated in the cooling system.

In an aircraft-engine cooling system the main coolant flow is from the pump to the engine, through the engine to a radiator, and through the radiator back to the pump. An expansion tank is located at the engine outlet. Two different circuits, the series circuit and the shunt circuit, are in current use. These circuits differ mainly in the manner in which the expansion tank is connected into the system. A schematic diagram of a series circuit is shown in figure 1(a). In this circuit the expansion tank is located in the main coolant line between the engine and the radiator. The shunt circuit is schematically shown in figure 1(b). The flow in this circuit differs from that of the series circuit in that only a small portion of the flow from the engine, about 1 or 2 percent, goes to the expansion tank. This flow through the expansion tank does not go through the radiator but returns directly to the pump inlet.

In any cooling system, the pressure in the expansion tank determines the absolute pressure level of the entire system. In a pressurized system the expansion-tank pressure is maintained higher than the atmospheric pressure by means of a pressure cap fitted to the tank. This pressure cap usually incorporates a pressure-relief valve to relieve excessive pressures and a vacuum or "sniffle" valve to protect the system against excessively low pressures. Because it is difficult to keep these valves and other piping connections from leaking slightly and because variations in flight conditions and engine power cause the coolant temperature to change, the highest pressure that can be reliably maintained in the expansion tank is the vapor pressure of the coolant at the block-outlet temperature.

The vapor pressure of the coolant at the pump inlet is lower than the vapor pressure in the expansion tank owing to the temperature drop across the radiator. This temperature drop is not very large (about 10° F); hence the decrease in coolant vapor pressure between the expansion tank and the pump inlet is small. This small change in coolant vapor pressure therefore limits the pressure proximity at the pump inlet to a relatively small amount. The absolute pressure at the pump inlet is equal to the expansion-tank pressure plus the differential-elevation head minus any pressure losses in the shunt line. It is therefore desirable to have as low a line pressure drop as possible between the expansion tank and the pump inlet. In this respect, the shunt system is better than the series system because the difference in pressure between the expansion tank and the pump inlet is not decreased by a pressure drop through the radiator and the main coolant piping. It is important, however, that the tank be so designed that the pressure loss in the shunt line does not become excessive.

The pump-inlet pressure may be increased by the use of special arrangements, such as applying heat to the expansion tank or by a venturi boost at the pump inlet, in which the shunt line from the expansion tank is connected to the throat of a venturi and the static pressure at the pump inlet is thereby increased by the amount of pressure recovery of the venturi. It should be noted, however, that the most desirable pressure proximity at the pump inlet is that value just sufficient to give satisfactory flow conditions. A system with an unnecessarily high pressure level is undesirable because of the difficulty of maintaining tight engine-jacket seals and piping connections.

As the engine power is varied, the temperature rise of the coolant through the engine, and hence the pressure proximity at the pump inlet, will vary for constant engine speed. If the pump is operating under cavitating conditions but still in a relatively stable region, the variation in the pressure proximity will result in a change of flow rate with a change in power; that is, an increase in power will raise the flow rate and a decrease of power will lower the flow rate.

APPARATUS

Engine Installation

The tests were conducted with an Allison V-1710-81 multicylinder engine mounted on a dynamometer stand equipped with a 2000-horsepower eddy-current dynamometer. Oil, refrigerated air, and atmospheric exhaust were supplied to the engine at specified conditions by auxiliary equipment.

Cooling Systems

System using unpressurized-type expansion tank. - Cooling system A, a conventional shunt type, used an expansion tank of the type in current use on unpressurized AN-E-2 ethylene glycol systems. Air-vapor separators were added to the system and two versions of their installation were tested: (a) when installed in the main coolant lines at the block outlets and (b) when installed in the shunt lines at the block outlets. Figure 2 is a schematic diagram of the system showing the air-vapor separators installed in the main coolant lines together with a cross-sectional drawing of the expansion tank. The version of the system with the air-vapor separators installed in the shunt lines at the block outlets is shown schematically in figure 3. A cross-sectional view of the expansion tank, which was modified for tests of this version of the system by the installation of a horizontal baffle inside the tank, is also shown in figure 3.

The system was constructed to simulate the installation in a typical pursuit-type military airplane with regard to length, size, and resistance of piping. The coolant flow divides at the pump outlet, follows parallel paths through each engine block and cooler, and returns to a Y connection at the pump inlet. The shunt circuit of the system consists of separate shunt lines at each block outlet, which join and enter the expansion tank, and a return line from the bottom of the expansion tank to the pump inlet.

The shunt lines from the block outlets, which were fitted with 1/8-inch orifices to restrict the flow, were joined together and connected to the expansion tank as shown in figures 2 and 3. The lines were mounted in a horizontal plane in order to decrease the possibility of air traps. The common line from the block shunt lines was extended halfway down into the tank and discharged tangentially along the inner wall. The vent lines from the vapor separators were connected to each side of the expansion tank. The return line from the bottom of the expansion tank was connected to the pump-inlet cover as shown in figures 2 and 3.

The expansion tank was mounted between the cylinder blocks and was fitted with an altitude-compensated pressure-relief valve set to relieve at 35 pounds per square inch absolute. The horizontal baffle, which was fitted to the tank for the tests of the second version of the system, was installed in an attempt to improve vapor and air separation from the liquid in the tank by shielding the inlet of the pump return line from the direct discharge of the vent lines. The main-line air-vapor separator shown in figure 4(a) is of the centrifugal type and has a ratio of inlet area to throat area of 3:1. The shunt-line air-vapor separator shown in figure 4(b) was made by

flattening a bend in the tubing to provide a reduction of area at the throat and adding a vent line to the inside radius of the bend. The Allison shrouded-impeller coolant pump fitted with the Y inlet cover was used. The pump speed is 1.234 times the engine speed. The coolant temperature was controlled by means of a water-cooled heat exchanger and an air-operated bypass valve installed in the water line. The normal pressure drop across the system from the pump outlet to the pump inlet was about 34 pounds per square inch at a flow rate of 250 gallons per minute.

The coolant-flow rate was determined by two similar venturi tubes installed in the main coolant lines of each block. System pressures were measured at the locations indicated in figures 2 and 3 by calibrated Bourdon-tube gages. Iron-constantan thermocouples connected to a self-balancing potentiometer were used to measure the coolant temperature at the cylinder-block outlets, at the expansion tank, and at the pump inlet from the right bank. Sight glasses installed in the main coolant lines, in the vent lines, and in the expansion tank return line, at the locations shown in figures 2 and 3, permitted observation of the coolant. A sight glass and an electric light were also installed in the baffled expansion tank to permit observation of the internal-flow conditions. Compressed-air lines were connected to the main coolant lines at the locations indicated for aerating the coolant in order to test the vapor separators and to control the pressure of the system at the desired levels.

System using Linde expansion tank. - The cooling system using the Linde expansion tank (system B) is the same as system A except that no main and shunt-line air-vapor separators were installed and the expansion tank is different. Figure 5 is a schematic diagram of system B. The expansion tank (figs. 6 and 7) consisted essentially of an outer or main tank inside of which was located a centrifugal air-vapor separator. This air-vapor separator received the flow from the block shunt lines and discharged it directly to the pump inlet. The outer expansion tank was also connected to the pump inlet by a static line. Because the flow from the block shunt lines did not enter the outer tank (except for the negligible flow from the separator vent), there was practically no flow in this static line. The air-vapor separator, which was at a higher pressure than the outer static tank, was provided with a restricted vent in order to obviate excessive flows into the outer tank and through the static line. Heat supplied by the hot coolant in the air-vapor separator maintained the temperature of the liquid in the outer tank at approximately the same temperature as the outlet coolant, thus assuring maximum coolant-vapor pressure in the tank.

In order to permit the installation of the expansion tank in the engine used in these tests it was necessary to fabricate a tank

at the Cleveland laboratory that was 2 inches shorter than the Bell production model of the Linde design. The same style pressure-relief valve was used as on the expansion tank of system A and a screen was installed, as shown in figure 6, to prevent dirt from passing into the valve and depositing on the neoprene seal.

Tests were conducted with two different sets of flow-restriction orifices in the block shunt lines. The first set was 1/8 inch in diameter and the second set was 3/16 inch in diameter. A compressed-air line and a blow-off cock were connected to the expansion tank at the location shown in figure 6.

The instrumentation was the same as for the tests of system A except for measurement of the coolant condition at the pump inlet and the flow rate through the shunt lines. Mercury manometers were connected to the pump suction lines from both banks and an additional thermocouple was so installed in the left bank suction line that the coolant temperature and pressure could be determined in both of the pump-suction lines. Manometers were fitted across each of the flow-restriction orifices in the shunt lines to measure the flow rate to the expansion tank. The complete shunt-line assembly was subsequently calibrated by a bench test.

Venturi-Calibration Setup

One of the venturi tubes used to measure the coolant-flow rate was calibrated by a bench test over the range of flow rates and pressure proximities encountered in the engine tests. A sight glass was installed immediately downstream from the venturi to permit observation of the flow. Both the flow rate and the upstream pressure on the venturi were controlled by valves located at the venturi entrance and downstream from the sight glass. Differential manometers were fitted across both the venturi and the sight glass; single-tube manometers were connected to the entrance of both the venturi and the sight glass to provide a complete pressure survey. The flow rate was determined by a weighing tank.

TEST PROCEDURE

Tests were conducted to determine coolant-flow rates for both systems over a range of engine speeds at constant pump-inlet pressure and over a range of expansion-tank pressures at constant engine speed. The following table gives a summary of the test conditions for both systems:

System	Engine power (bhp)	Engine speed (rpm)	Pump-inlet pressure (lb/sq in. abs.)	Flow-restriction orifice in shunt line (in.)
Variable engine speed				
A	Constant throttle (635 to 725)	2000 to 3000	27.9	1/8
B	725 ±35	2000 to 3000	27.0	1/8
			28.0	1/8
			28.0	3/16
Variable expansion-tank pressure				
A	1035 ±5	3000 ±5	26.1 to 32.8	1/8
B	735 ±5	3000 ±5	23.2 to 31.2	1/8

The coolant-outlet temperature was held at 250° ±2° F and the carburetor-mixture control was maintained in automatic-rich position for all runs.

For the tests of system A, the expansion-tank pressure was increased by the introduction of compressed air into the system at the main block outlet line; coolant was discharged through a bleed-off valve in order to decrease the pressure. During the tests of system B the expansion-tank pressure was increased by admitting compressed air into the tank and decreased by bleeding-off vapor from the tank through a blow-off valve. For the variable engine-speed runs of both systems the pump-inlet pressure was maintained constant by adjusting the expansion-tank pressure.

The pump-inlet pressure taps were located about 1 foot upstream from the pump suction to avoid error from the uneven pressure distribution in the long-radius elbows. The measurements of pump-inlet pressure thus obtained were subsequently corrected for this line loss and therefore the data presented herein represent the pressure immediately upstream from the Y suction connection of the coolant pump.

RESULTS AND DISCUSSION

Effect of engine speed on coolant-flow rate. - The variation of coolant-flow rate with engine speed for system A at a pressure proximity (static pressure minus vapor pressure) of 7 pounds per square inch at the pump inlet and for both types of air-vapor separator is shown in figure 8(a). Similar data for system B for two values of

the pressure proximity at the pump inlet and with two different sized orifices in the shunt lines are shown in figure 8(b). The data points of both systems have been corrected for slight variations in pressure proximity encountered in the tests.

Figure 8(a) shows that the coolant-flow rate of system A varies directly with engine speed up to nearly 3000 rpm, indicating that the pump operates almost free of cavitation at a pressure proximity of 7.0 pounds per square inch. A slightly higher flow rate, particularly at the higher engine speeds, is noted for system A with the air-vapor separators installed in the block shunt lines. The loss in flow rate caused by decreasing the pressure proximity from 6.0 to 5.0 pounds per square inch, shown in figure 8(b), is greater at the higher engine speeds than at the lower engine speeds because the higher velocity of the liquid through the pump results in greater entrance pressure losses, which increases the tendency to vapor formation. For a pressure proximity of 6.0 pounds per square inch (corresponding to a pump-inlet pressure of 28.0 lb/sq in. absolute), the coolant-flow rate varies directly with engine speed up to 2600 rpm, indicating the pump to be operating free from cavitation in this region; whereas for a pressure proximity of 5.0 pounds per square inch (corresponding to a pump-inlet pressure of 27.0 lb/sq in. absolute), a slight amount of cavitation is noted at all engine speeds above 2000 rpm. The change from 1/8-inch to 3/16-inch orifices in the shunt lines of system B has no apparent effect on the coolant-flow rate.

Effect of expansion-tank pressure on coolant-flow rate. The variation of coolant-flow rate with expansion-tank pressure is shown in figures 9(a) and 9(b) for systems A and B, respectively. The data for both types of vapor separator used in system A fall on one curve, indicating no significant difference in system operation between the two types of vapor separator. A maximum coolant-flow rate of about 275 gallons per minute was obtained for both systems at an expansion-tank pressure of about 32 pounds per square inch absolute. The coolant-flow rate falls off more rapidly with a decrease in expansion-tank pressure for system A, however, than for system B. The temperature of the liquid in the expansion tank was generally about 1° F lower than the block-outlet temperature. Thus, for a coolant-outlet temperature of 250° F the vapor pressure in the expansion tank is about 26 pounds per square inch absolute, resulting in a coolant-flow rate of about 200 gallons per minute for system A and of about 255 gallons per minute for system B. At this point the curve for system A is almost vertical, which indicates that a further slight reduction in expansion-tank pressure due to leaks in the tank or in other parts of the system would probably cause a breakdown of coolant flow. For system B, however, the flow-rate curve is sufficiently flat in this region to indicate stable and safe operation.

Relation between pump-inlet pressure and expansion-tank pressure. - Figure 10 shows the relation between pump-inlet pressure and expansion-tank pressure for systems A and B. Because the expansion tank and the pump inlet are connected by a return line, or static line, as shown in figures 2, 3, and 5, the pump-inlet pressure is equal to the expansion-tank pressure plus the differential-elevation head minus the line pressure drop. For any fixed expansion-tank pressure the pump-inlet pressure is seen to be about 6.0 pounds per square inch higher in system B than in system A, which results in a pump-inlet pressure of about 27.5 pounds per square inch absolute in system B as compared with a 21.5 pounds per square inch absolute pump-inlet pressure in system A when normal (approximately 26 lb/sq in. absolute) coolant vapor pressure exists in the expansion tank. This difference is due to the fact that the shunt-line flow in system A goes through the return line connecting the expansion tank to the pump inlet with attendant line pressure losses, whereas in system B the pump inlet is connected to the expansion tank by the static line in which there is very little flow. The pressure losses in the return line from the air-vapor separator of system B does not decrease the pump-inlet pressure because the pressure within the separator is correspondingly higher than the expansion-tank pressure. The differential-elevation head between the pump and the expansion tank is the cause of the pump-inlet pressure in system B being about 1.5 pounds per square inch higher than the expansion-tank pressure.

Variation of coolant-flow rate with pump-inlet pressure. - Figures 11(a) and 11(b) show the variation of coolant-flow rate with pump-inlet pressure for systems A and B, respectively. A corresponding variation of flow rate is obtained when it is plotted against the pressure proximity at the pump inlet, thereby giving the cavitation characteristics of the pump. (See fig. 12.) The cavitation curves obtained for both systems are similar and show that full liquid flow is not obtained until a pressure proximity of about 8 pounds per square inch is reached. As noted on the curves, when coolant vapor pressure exists in the expansion tank, the pressure proximity at the pump inlet is about 5 pounds per square inch higher for system B than for system A. (The difference in the pressure proximity between the two systems is not so great as the difference in the pump-inlet pressure owing to the lower engine power for the tests of system B.) This difference in pressure proximity, for the same expansion-tank pressure, results in a greater degree of pump cavitation and a correspondingly lower flow rate for system A than for system B, as was noted in figure 9.

The system resistance did not remain constant during these tests but increased about 25 percent as the pressure in the system was reduced. This increase was due to an effect introduced by the venturi

in the pump suction line and probably caused the flow rate to start to fall off at a higher pressure than would the curves of flow rate for a constant system resistance. The cause of this variation in system resistance and its effect on the performance of the system will be discussed in greater detail later in this report.

Flow rate through the shunt lines. - The effect of engine speed on the flow rate through the shunt lines of system B is shown in figure 13(a). A relatively small flow rate, which increases with engine speed, is noted for both sizes of flow-restriction orifices.

Figure 13(b) shows the effect of pump-inlet pressure on the flow rate through the shunt lines. The pressure drop across the shunt-line system is the difference between the block-outlet pressure and the pump-inlet pressure; the small effect of pump-inlet pressure on the flow rate is caused by an increase in the pump-inlet pressure accompanied by an almost equal increase in the block-outlet pressure.

Effect of coolant-flow rate on pressure drop. - The pressure drops both through the entire system (pump outlet to pump inlet) and from the block outlet to the pump inlet are shown as a function of the total coolant-flow rate in figure 14(a) for system A and in figure 14(b) for system B. For the variable-speed runs, the pressure drop increases with the square of the flow rate in the normal manner and the agreement between the two systems is very good. The pressure drop for the variable-pressure runs, however, is seen to remain constant in system B over nearly the full range of flow rates tested; for system A, the pressure drop remains constant over a considerable range of flow rates and gradually decreases as the low flow rates are reached. Inspection of the data showed that, for each system at the point of intersection, the pump-inlet pressure for the variable-speed curve is equal to that for the variable-pressure curve.

This unusual variation of pressure drop with flow rate during the variable-pressure runs may be explained by the results of the venturi-calibration tests. During the calibration tests at low pressures, large quantities of vapor were observed in the sight glass, indicating that considerable flashing or boiling of the coolant occurred in the venturi. As the upstream pressure on the venturi was increased, the flashing was suppressed until finally full liquid flow was obtained. The pressure survey through the venturi and sight glass for both flashing and nonflashing runs is given in figure 15. For clarity, only a few typical runs are presented. For the flashing runs, the upstream pressure was adjusted to a value that resulted in reconversion of two-phase flow (vapor and liquid) to full liquid flow in the downstream portion of the sight glass. It is noted that for the

flashing runs the pressure recovery did not take place in the downstream cone of the venturi but occurred, to a limited extent (about one-fourth of that obtained in nonflashing runs), in the region where the reconversion to liquid flow occurred. This result is attributed to boundary separation in the venturi cone and the attendant eddy losses. The higher over-all pressure loss for the flashing runs is shown in figure 16 in which the pressure drop from the venturi entrance to the downstream end of the sight glass is plotted against the flow rate. Figure 17 gives the venturi calibration for both flashing and nonflashing runs. In spite of this wide variation in flow conditions, all data points are seen to be within 8 percent of the calculated calibration curve.

This higher over-all pressure loss through the venturi and downstream piping when falshing of the coolant occurred affords the explanation of the unusual variation of pressure drop with flow rate noted in figure 14. As the coolant-flow rate through the cooling systems was reduced by decreasing the pump-inlet pressure, vaporization occurred in the venturi, which increased the resistance of the system. This increase in system resistance, together with the decreased flow rate, resulted in a constant over-all pressure drop over a considerable range of flow rates. At the lower flow rates encountered in the tests of system A, the pressure drop varies with flow rate in the normal manner but is higher than the variable-speed pressure-drop curve. It is believed that in this region the vaporization in the venturi reached a maximum and did not contribute further increase to the system resistance.

Effect of venturi on performance of the cooling system. - From the foregoing considerations, it should be noted that actual aircraft installations of the cooling systems (that is, without the flow-measuring venturi) may be expected to perform with somewhat higher flow rates at low pump-inlet pressures than is shown by the test results presented herein because the effective system resistance will not be increased by vaporization in the venturi. Inspection of the data indicates that the maximum increase in system resistance caused by vaporization in the venturi amounted to about 25 percent of the normal resistance and occurred at a pump-inlet pressure of about 24 pounds per square inch absolute (pump-inlet pressure proximity of 2 lb/sq in.). At this point the coolant-flow rate without the venturi in the system would be expected to be about 12 percent higher than obtained in the tests. Inasmuch as the pump-inlet pressure is controlled by the shunt-line branch of the system, the pressure proximity at pump inlet is not affected by the presence of the flow-measuring venturi in the main coolant line. System A, which was found to be operating with a pressure proximity of less than 1 pound per square inch at the pump-inlet (see fig. 12(a)) when

coolant-vapor pressure existed in the expansion tank, would therefore still be operating in an unstable region even with the venturi removed.

System operating experience. - Both systems were found to be difficult to fill after they had been completely drained. In order to fill the systems completely, it was necessary first to fill as much as possible, run the engine for a short period, and then shut down and add more coolant.

Although the expansion tanks were vented to the atmosphere during filling, this difficulty was not completely eliminated because the expansion tank is located at a lower level than the engine blocks. It was therefore also necessary to vent the engine blocks to the atmosphere. Connections for both of these vents have been provided in a new type filler cap designed by the Bell Aircraft Corporation. Both of these vents are closed off when filling is completed in order to prevent circulatory flow of coolant from the filler cap into the tank during operation and air from being drawn into the engine blocks after a shutdown.

The effectiveness of the air-vapor separators was tested by noting the removal of air and vapor during the warm-up period and by introducing air into the main coolant stream at both block outlets immediately upstream of the sight glasses. The main-line separators of system A removed the large air and vapor bubbles in less than 1 minute and completely removed small bubbles in fine suspension in about 5 minutes. The shunt-line separators required about 10 to 15 minutes to remove all the air and vapor. The internal air-vapor separator of the expansion tank of system B did not give as rapid separation as the main-line separators used in system A but did provide quicker separation than the shunt-line separators.

For both systems small bubbles of air or vapor were observed at all times in the return line from the expansion tank to the pump inlet, which indicates a continual pickup of vapor from the expansion tank into the system. The addition of the horizontal baffle to the expansion tank of system A decreased the amount of vapor pickup from the tank but did not completely eliminate this condition. Vapor was also present in the static line of the system B tank, which indicates that the liquid in the tank was fairly well agitated and mixed with the vapor. This condition was also noted during blow-off of excess pressure in the tank. Although the blow-off valve was connected to the upper section of the tank, a considerable quantity of liquid was discharged at every blow-off. In order to ameliorate this condition of vapor pickup, the Linde Air Products Company have recently added a shield over the end of the

vapor-separator vent line in such a way that the stream issuing from it will not agitate the liquid in the tank. It should be noted that the outlet of this vent line should be below the level of the liquid to prevent air from being drawn into the system after a shutdown.

A slight leakage, which was encountered through the pressure-relief valve at all times during the tests of system A, made it difficult to maintain high expansion tank pressures. The addition of the screen under the pressure-relief valve of the system B tank, which was installed in an attempt to eliminate the leakage, became so clogged with solid material that it prevented the pressure-relief valve from operating properly and was consequently removed.

SUMMARY OF RESULTS

The results of the engine tests described herein on two pressurized shunt-type cooling systems using a 30-70 glycol-water mixture as coolant indicate that the main difference in the performance of the two cooling systems is that, for any fixed expansion-tank pressure, the pump-inlet pressure obtained in system B (using Linde expansion tank) is about 6 pounds per square inch higher than that obtained in system A (using unpressurized-type expansion tank). This characteristic of the expansion tank used in system B is considered to be its principal advantage inasmuch as the higher pump-inlet pressures result in a high pressure proximity at the pump inlet and attendant stable operation of the pump at relatively low expansion-tank pressures and maximum system pressures. In current practice the highest expansion-tank pressure that can be reliably maintained is the coolant-vapor pressure corresponding to the block-outlet temperature.

The data obtained in these tests show that, when a coolant-vapor pressure of 26 pounds per square inch absolute (corresponding to a block-outlet temperature of 250° F) existed in the expansion tank, a coolant-flow rate of approximately 200 gallons per minute was obtained in system A and a coolant-flow rate of about 255 gallons per minute in system B. Although both of these flow rates are adequate to cool liquid-cooled aircraft engines at present rated powers, at this condition in system A the pump is operating in a dangerous and unstable region. The expansion-tank pressure curve for system A shows that a drop of 1/2 pound per square inch below the value corresponding to the vapor pressure in the expansion tank would result in a large decrease in coolant-flow rate, if not in a complete breakdown of the flow. This condition would also be present if the flow-measuring venturi were not included in the system because, as noted previously herein, the pump-inlet pressure is not influenced by the pressure drop through the venturi.

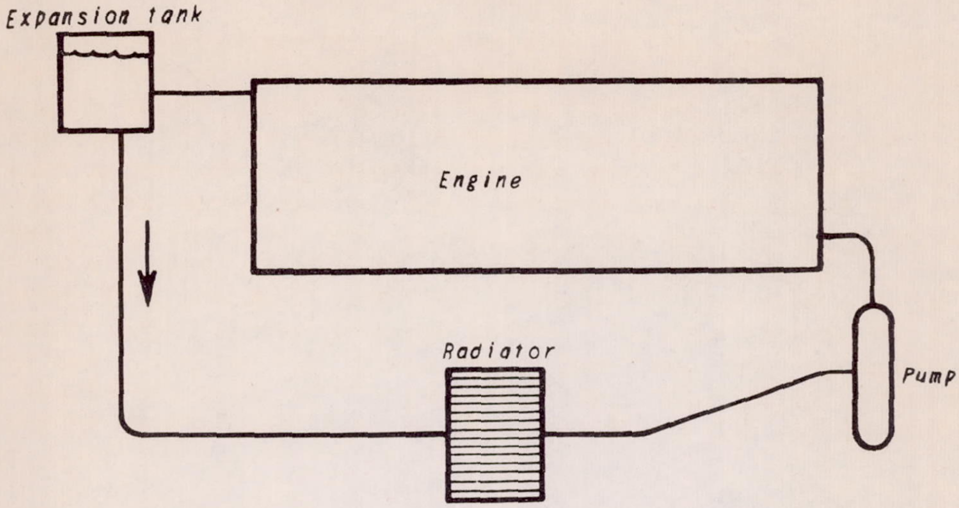
When coolant-vapor pressure existed in the expansion tank of system B, however, the pump was operating in a stable region and, although a slight amount of cavitation occurred, no harmful effects are expected. Higher pump-inlet pressures might be obtained for the same expansion-tank pressure by the use of a pump-inlet venturi or other boost arrangement. These devices are considered neither necessary nor desirable for present purposes when 30-70 glycol-water mixtures are used, in view of the negligible improvement in cooling resulting from the small possible increase in flow rate and the attendant higher system pressure levels. In this connection, it should also be noted that pressures higher than those necessary to give stable operation of the pump are undesirable owing to the difficulty of maintaining tight piping connections and engine-jacket seals.

Aircraft Engine Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio.

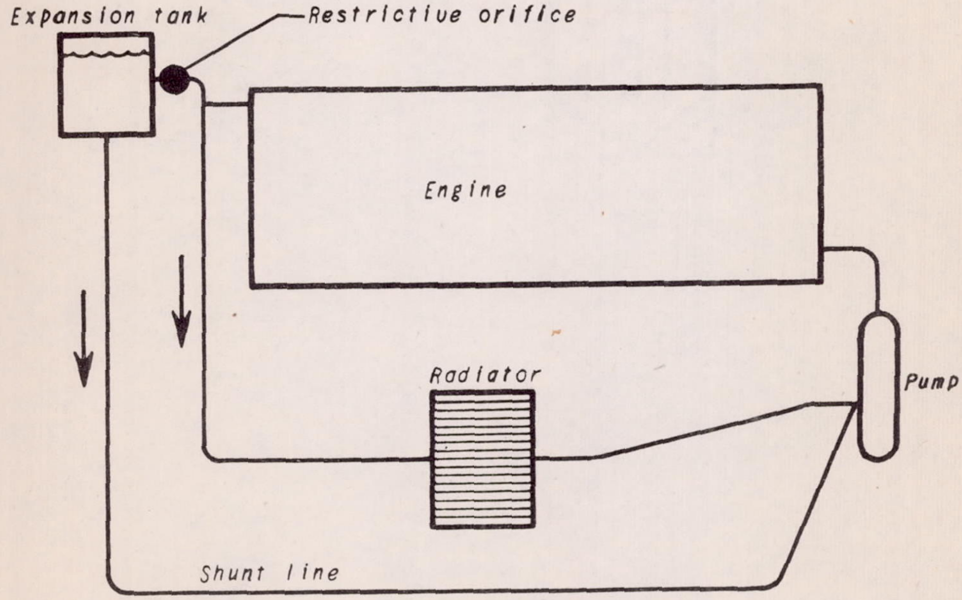
REFERENCES

1. Ellor, James E.: A Brief Survey of the Principles of Pressure Water Cooling. SAE Jour. (Trans.), vol. 51, no. 3, March 1943, pp. 65-68, 77.

2. Stamm, C. A., and McCravey, W. E.: Improvements in Cooling Systems. SAE Jour. (Trans.), vol. 52, no. 7, July 1944, pp. 34-35.



(a) Series-type circuit.



(b) Shunt-type circuit.

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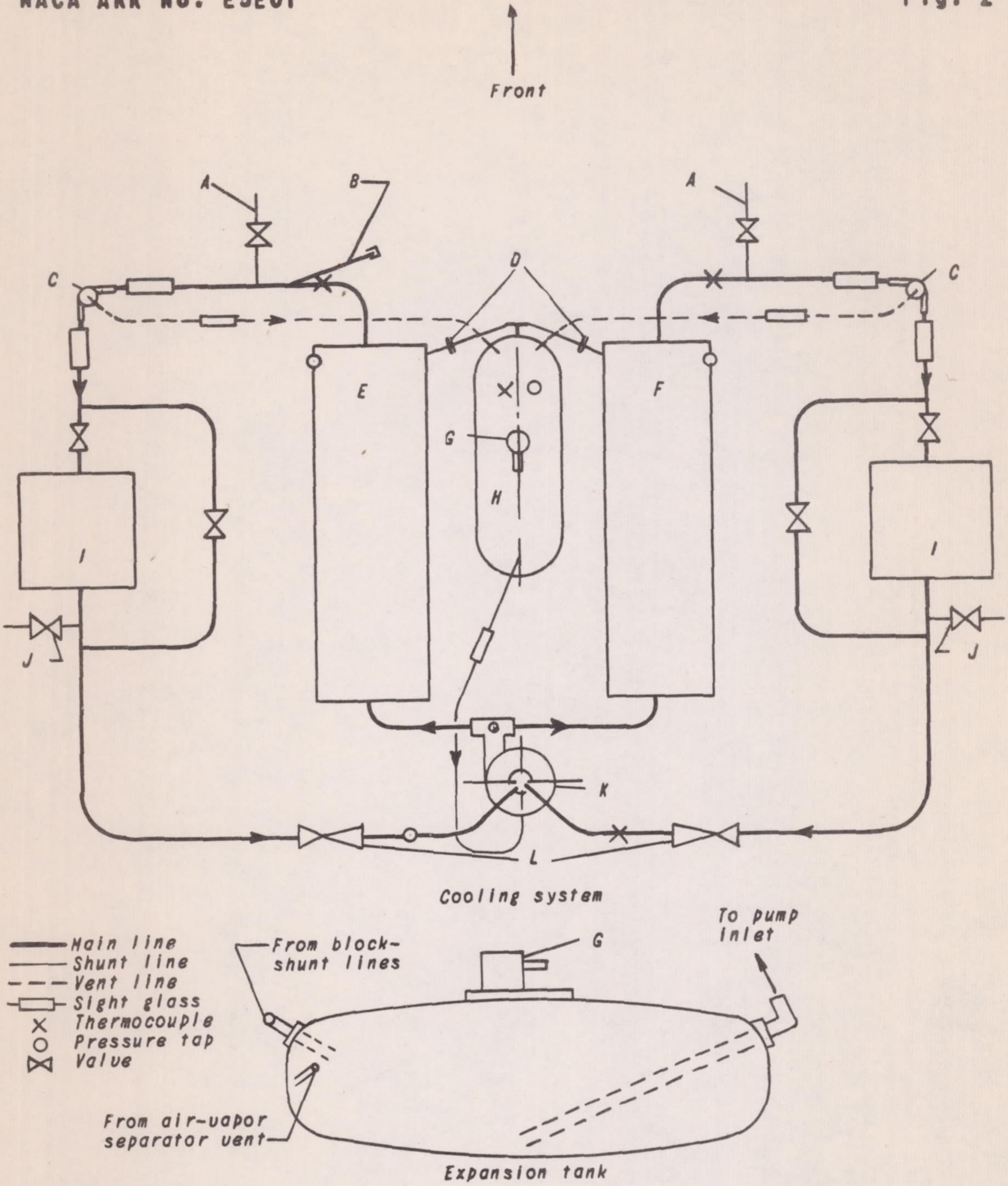
Figure 1. - Schematic diagram of basic types of cooling circuit.



A	Compressed air	E	Left bank	I	Cooler
B	Filler line	F	Right bank	J	Bleed valve
C	Air-vapor separators	G	Pressure-relief valves	K	Pump
D	Orifices	H	Expansion tank	L	Venturis

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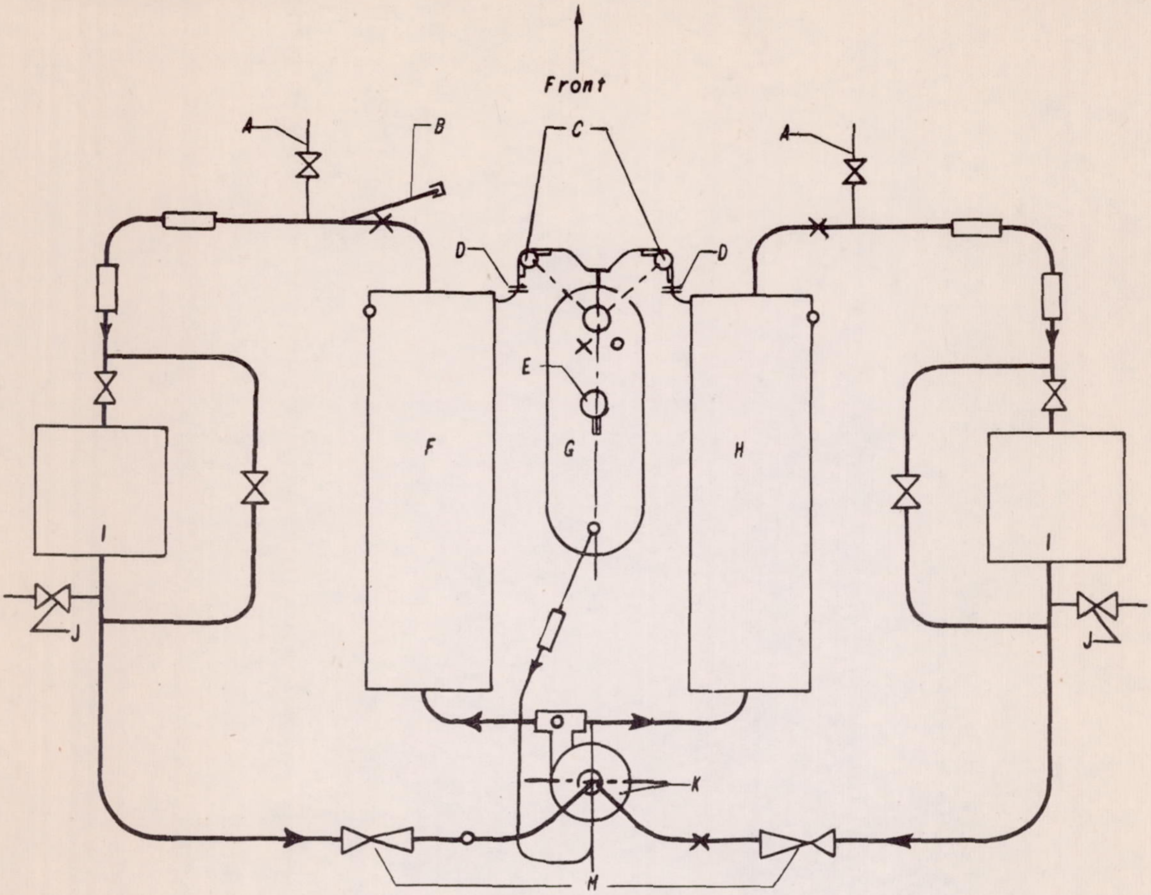
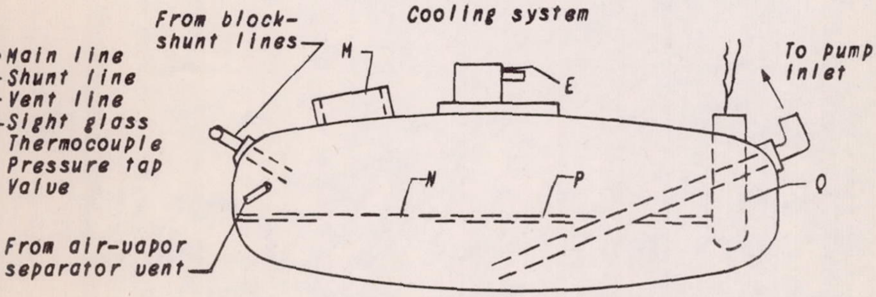
Figure 2. - Diagrammatic sketch of cooling system A showing air-vapor separators installed in main coolant lines and a cross-sectional view of the expansion tank.



Cooling system



- Main line
- Shunt line
- Vent line
- Sight glass
- × Thermocouple
- ○ Pressure tap
- Valve

Expansion tank

A Compressed air
B Filler line
C Air-vapor separators
D Orifice
E Pressure-relief valves
F Left bank
G Expansion tank
H Right bank
I Coolers
J Bleed valves
K Pump
L Venturis
M Sight hole
N Sight glass
P Baffle
Q Light

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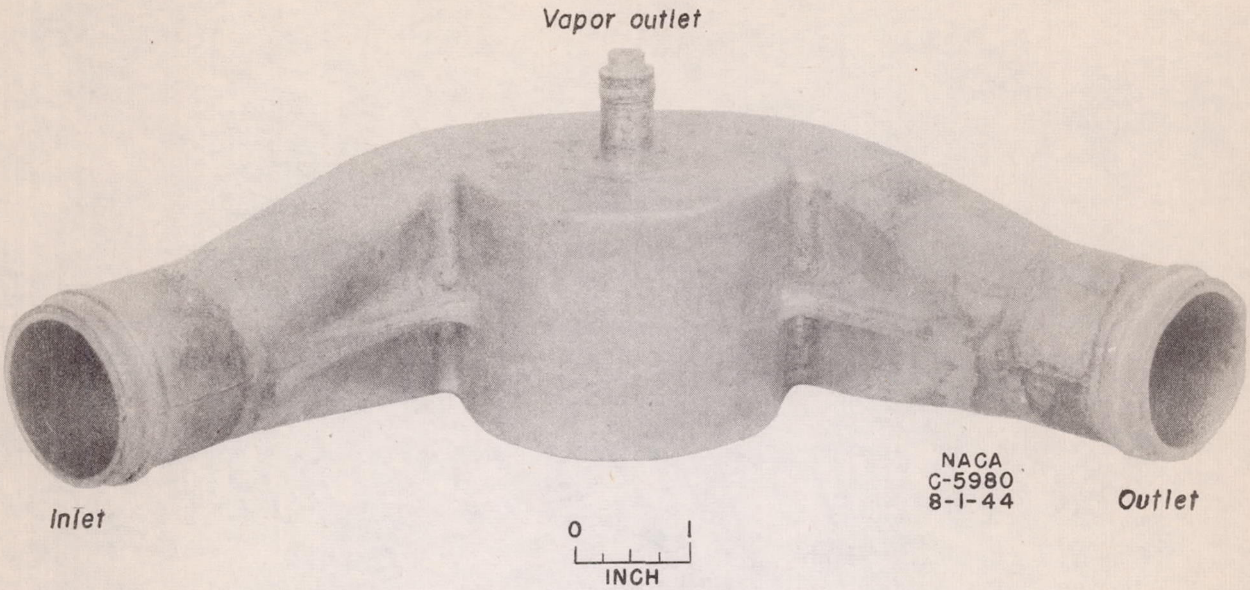
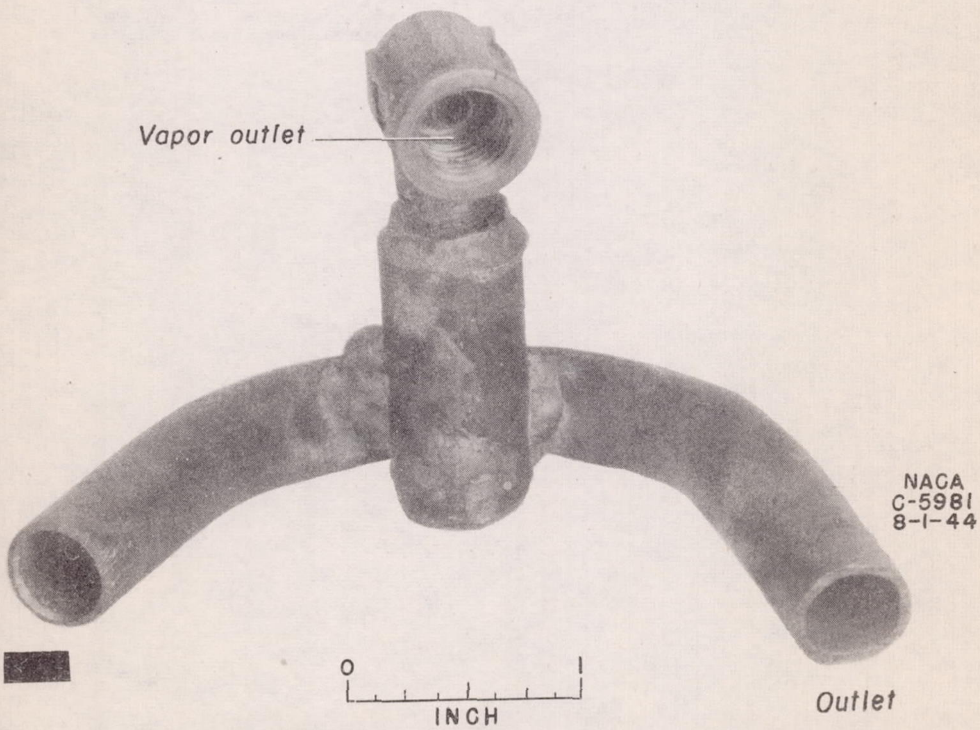
Figure 3. - Diagrammatic sketch of cooling system A showing air-vapor separators installed in the shunt lines and a cross-sectional view of baffled expansion tank.



(a) In the main line.



(b) In the shunt line.

Figure 4. - Vapor separators used in cooling system A.

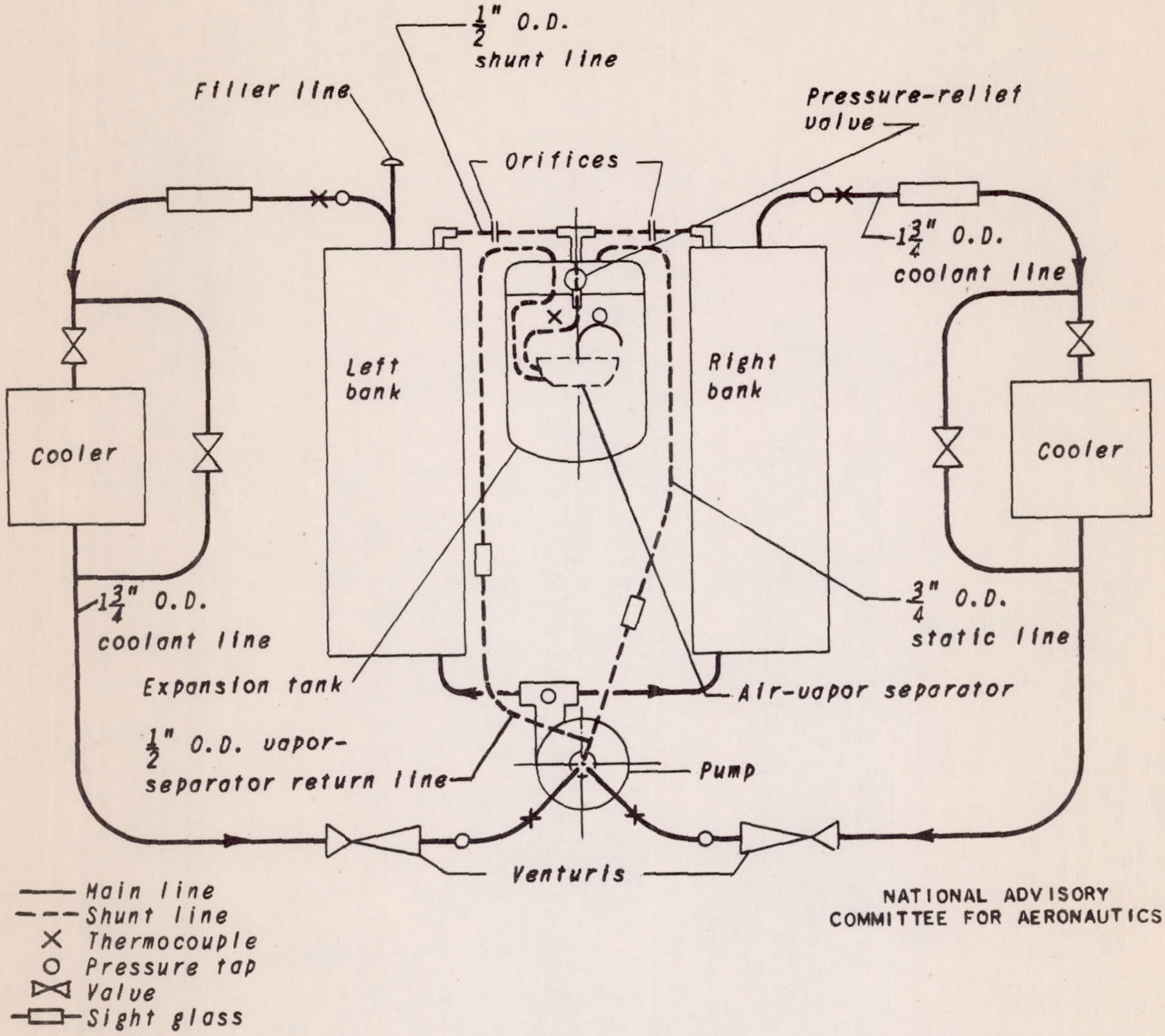


Figure 5. - Schematic diagram of cooling system B using Linde expansion tank.

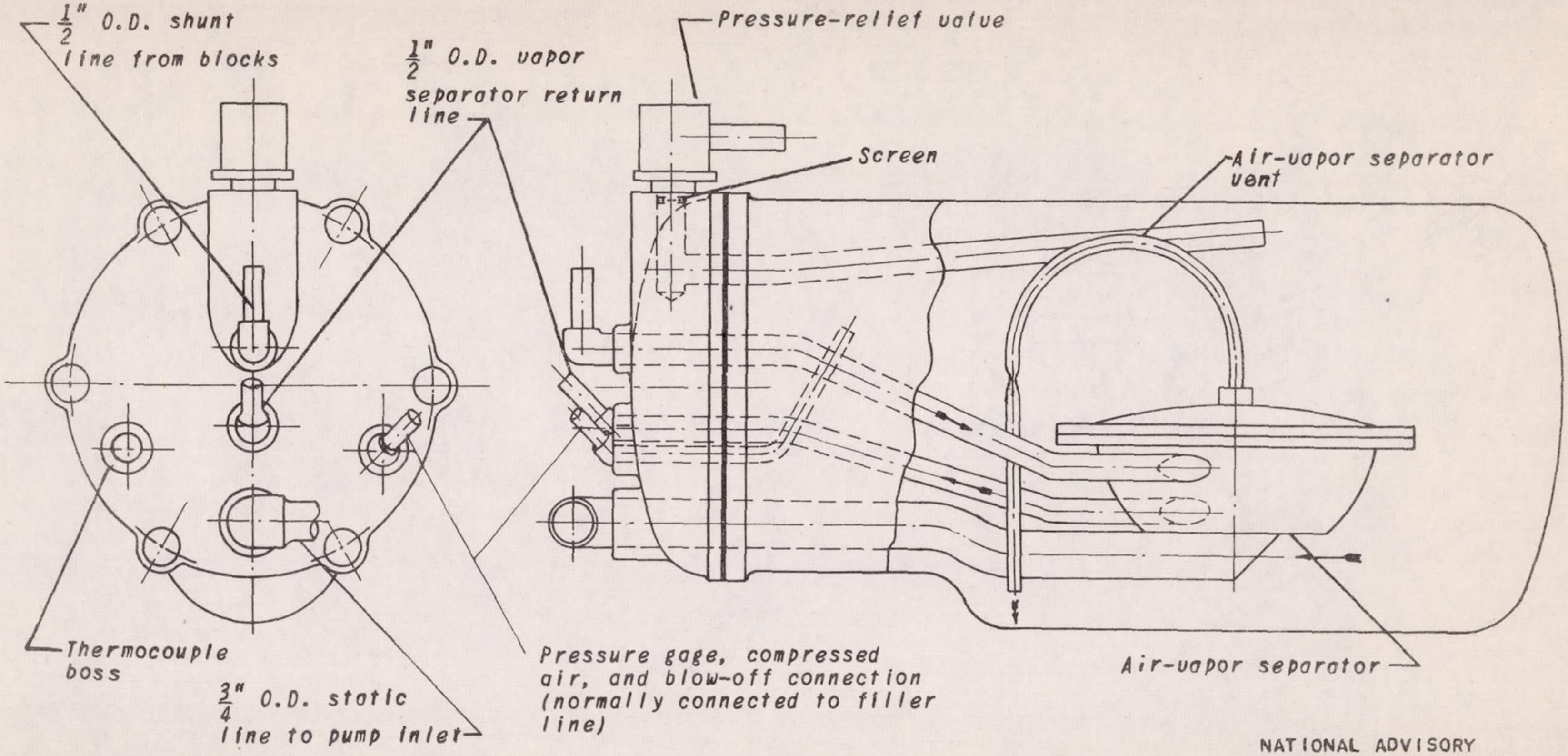


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Figure 6. - Details of Linde expansion tank used in cooling system B.

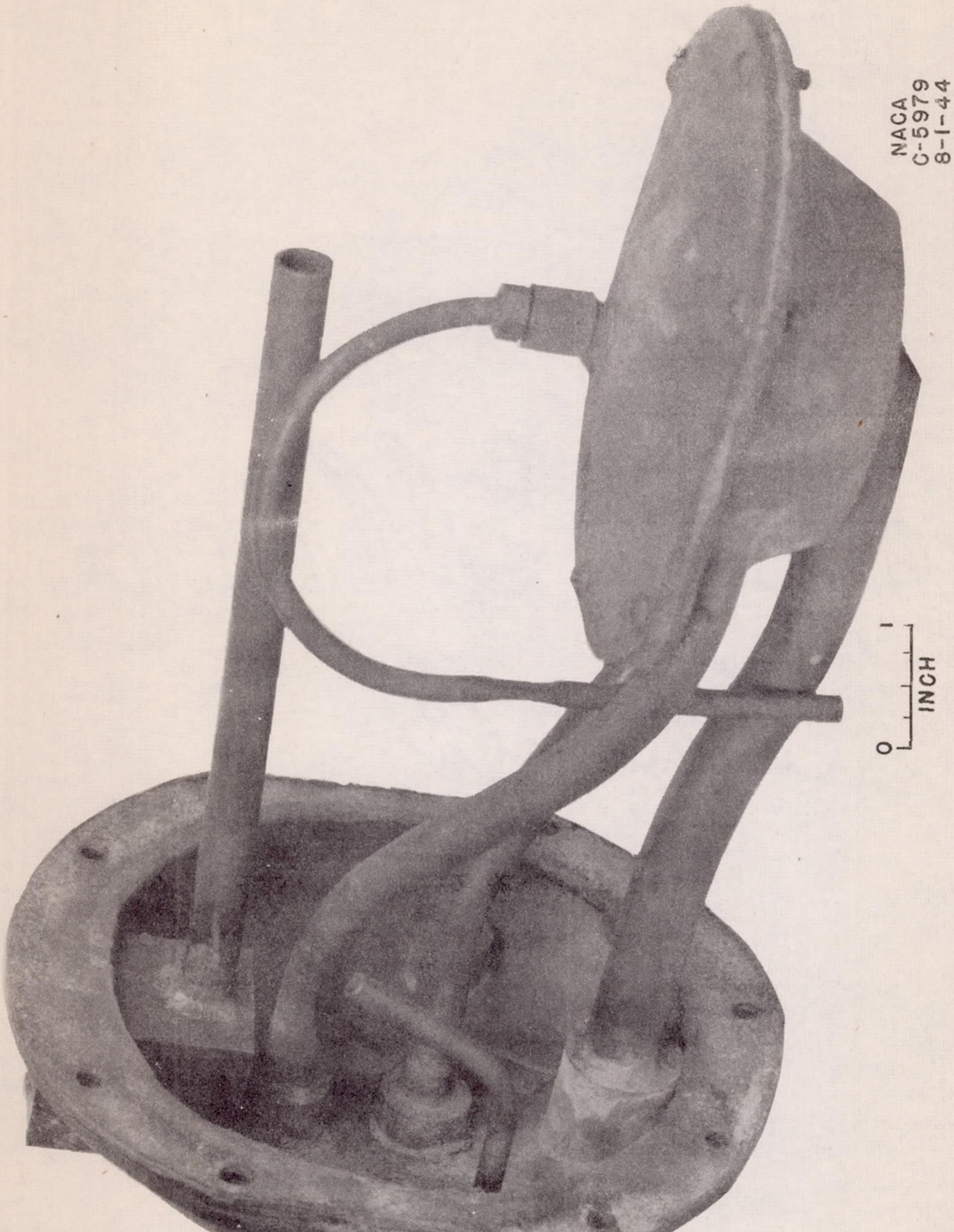
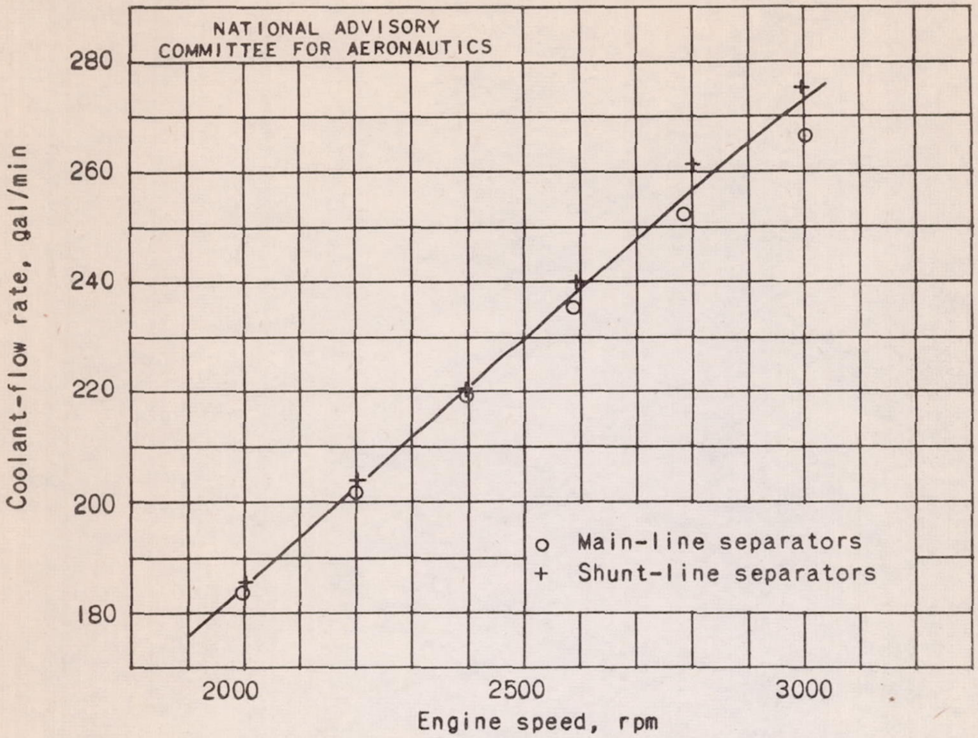
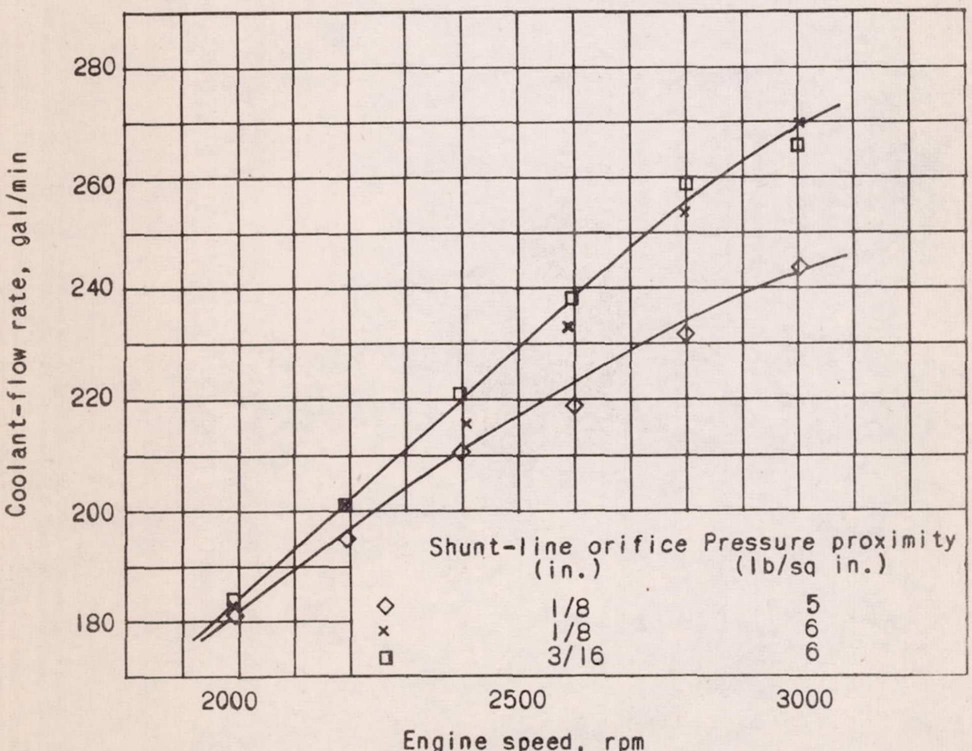


Figure 7. - Internal details of Linde expansion tank used in cooling system B.

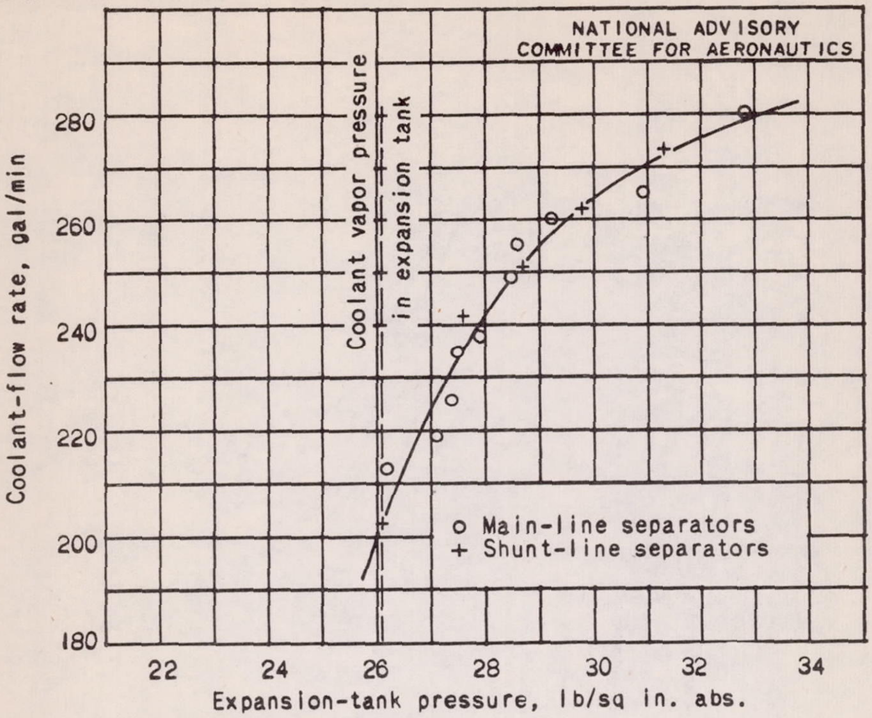


(a) Cooling system A Pressure proximity, 7 pounds per square inch.

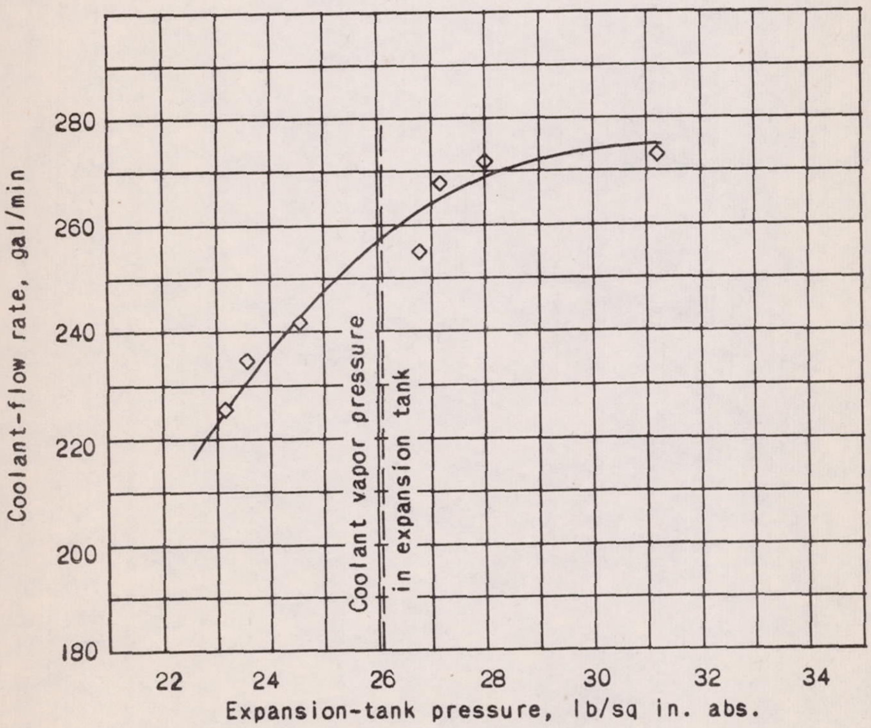


(b) Cooling system B.

Figure 8. - Variation of coolant-flow rate with engine speed. Coolant-out temperature, 250° F; coolant, 30 percent AN-E-2 ethylene glycol plus 70 percent water; pump speed, 1.234 times engine speed.



(a) Cooling system A.



(b) Cooling system B.

Figure 9. - Variation of coolant-flow rate with expansion-tank pressure. Engine speed, 3000 rpm; coolant-out temperature, 250° F; coolant, 30 percent AN-E-2 ethylene glycol plus 70 percent water; 1/8-inch shunt-line orifices.

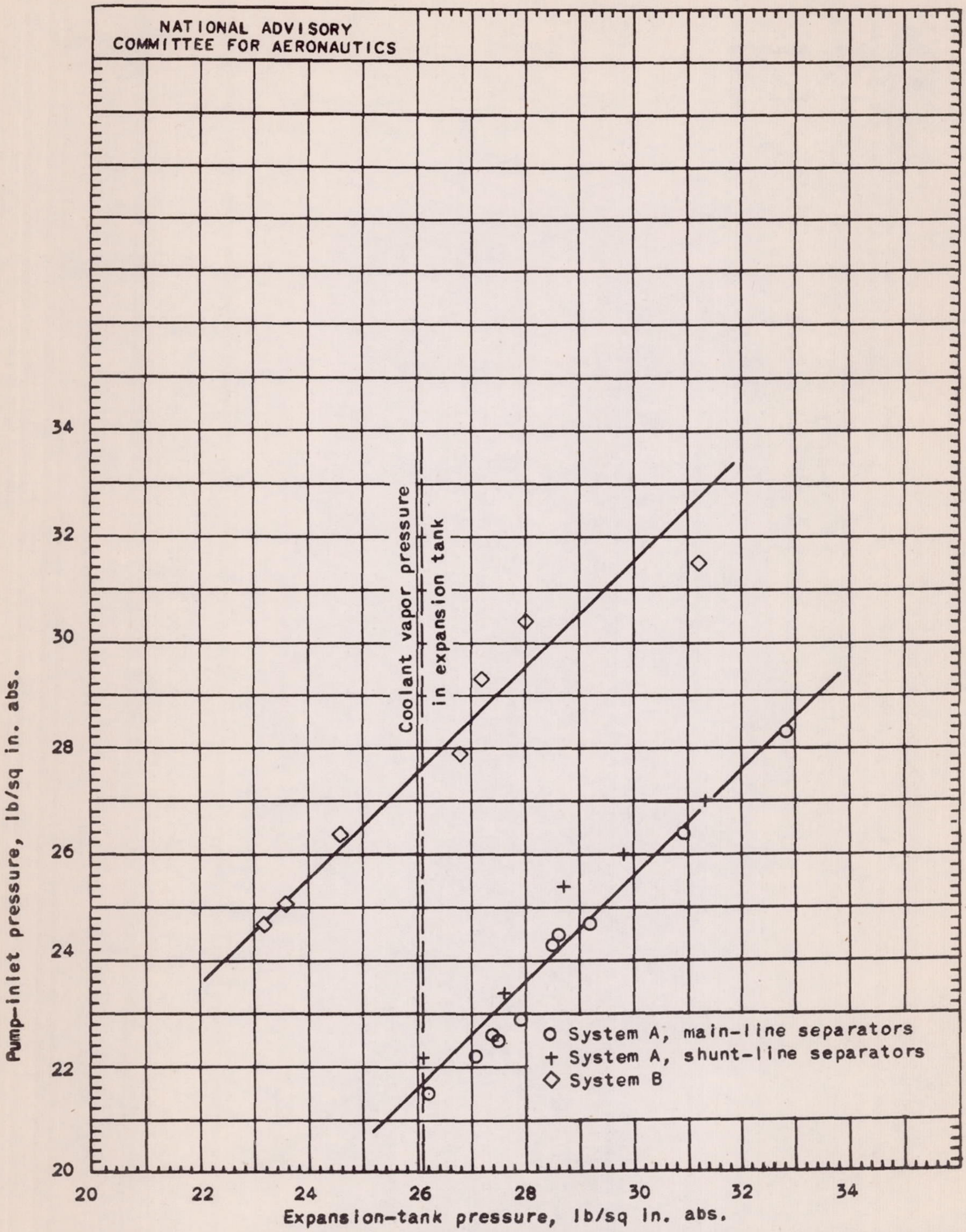
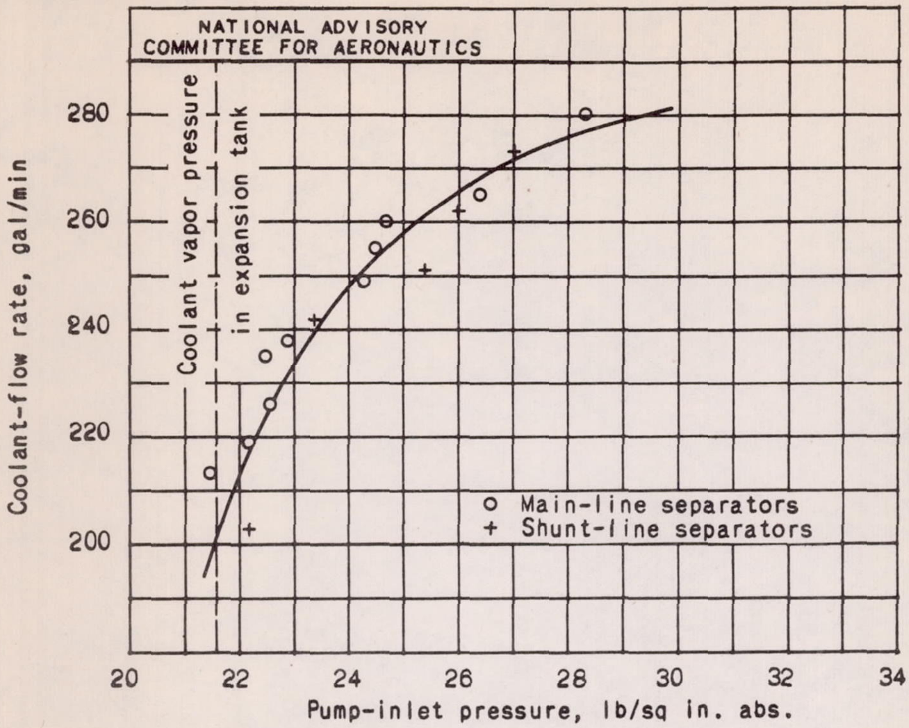
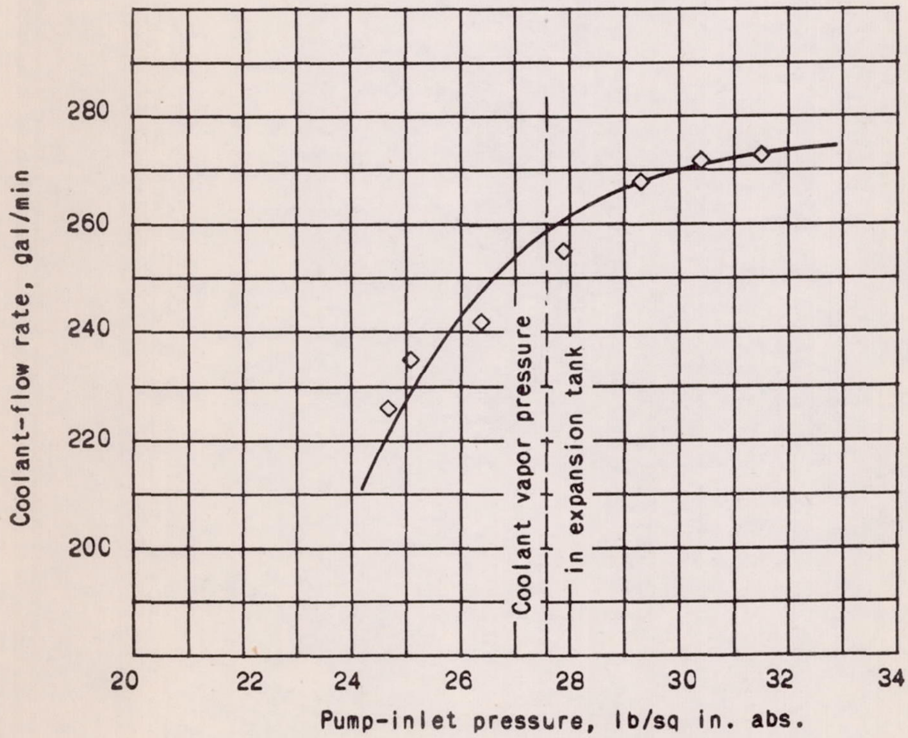


Figure 10. - Variation of pump-inlet pressure with expansion-tank pressure for both cooling systems A and B. Engine speed, 3000 rpm; coolant-out temperature, 250° F; coolant, 30 percent AN-E-2 ethylene glycol plus 70 percent water; 1/8-inch shunt-line orifices.

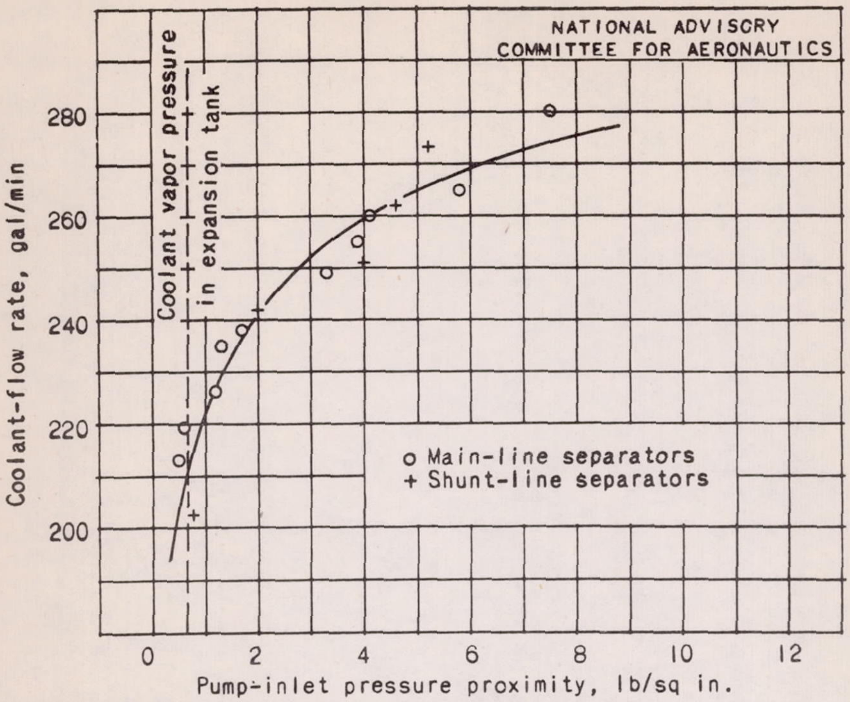


(a) Cooling system A.

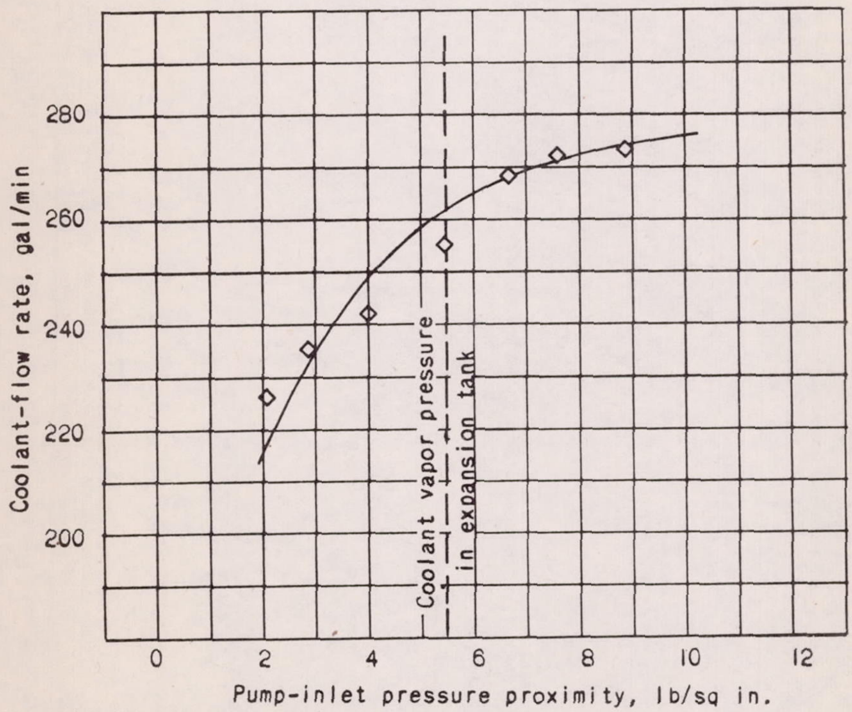


(b) Cooling system B.

Figure 11. - Variation of coolant-flow rate with pump-inlet pressure. Engine speed, 3000 rpm; coolant-out temperature, 250° F; coolant, 30 percent AN-E-2 ethylene glycol plus 70 percent water; 1/8-inch shunt-line orifices.

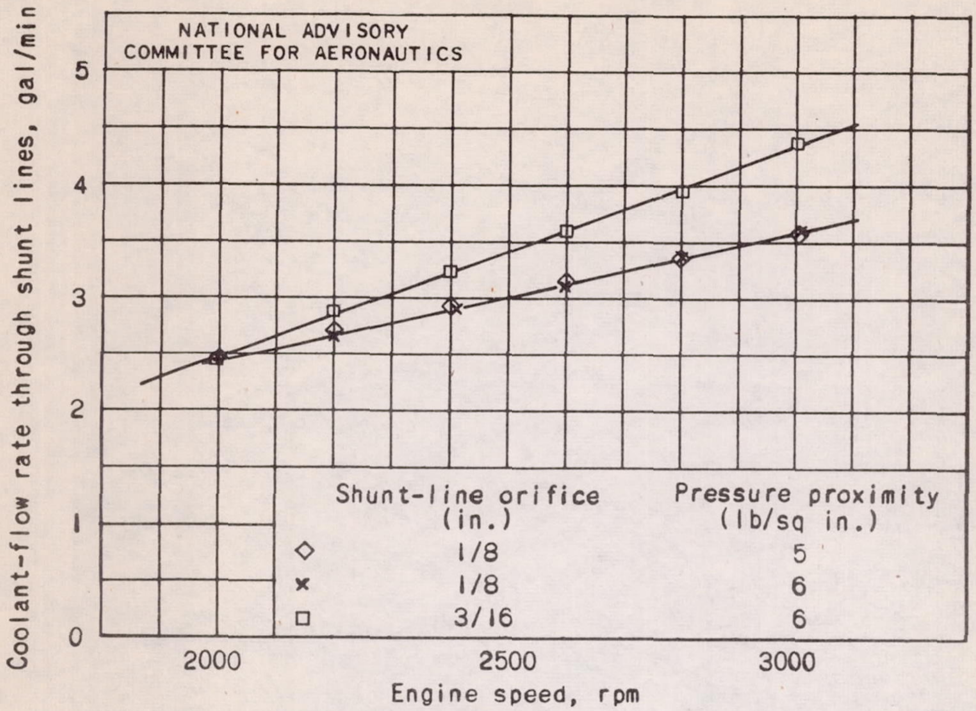


(a) Cooling system A.

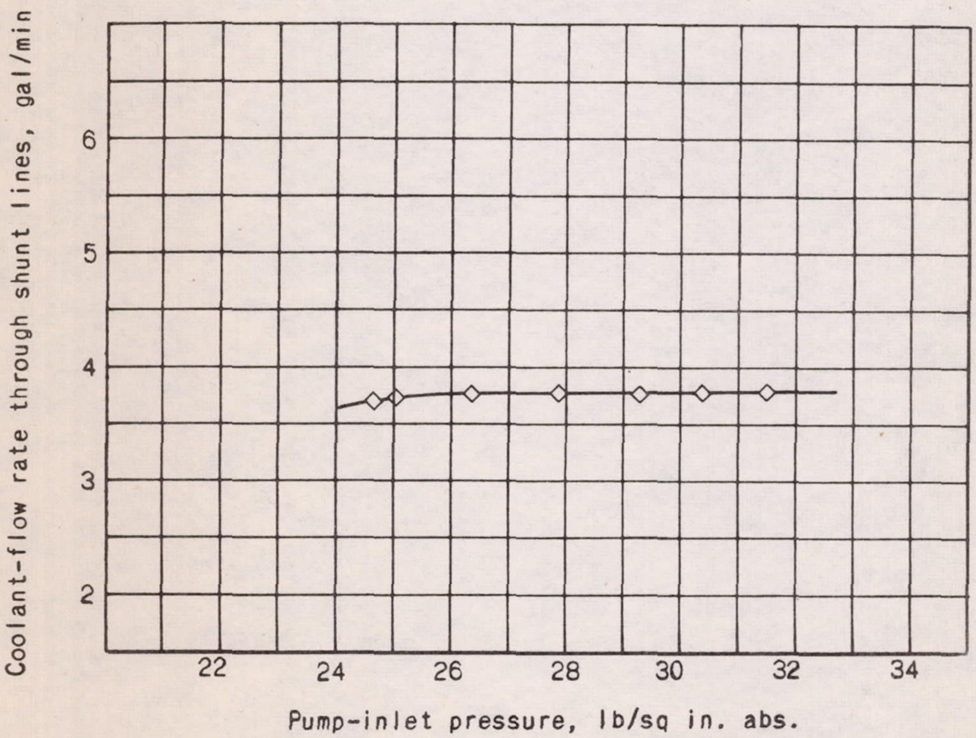


(b) Cooling system B.

Figure 12. - Variation of coolant-flow rate with pump-inlet pressure proximity. Engine speed, 3000 rpm; coolant-out temperature, 250° F; coolant, 30 percent AN-E-2 ethylene glycol plus 70 percent water; 1/8-inch shunt-line orifices.

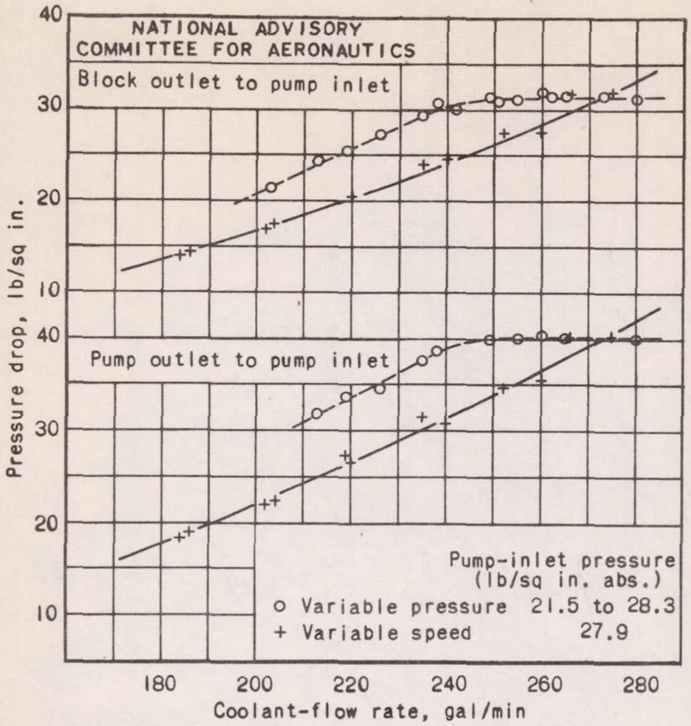


(a) Variation with engine speed.

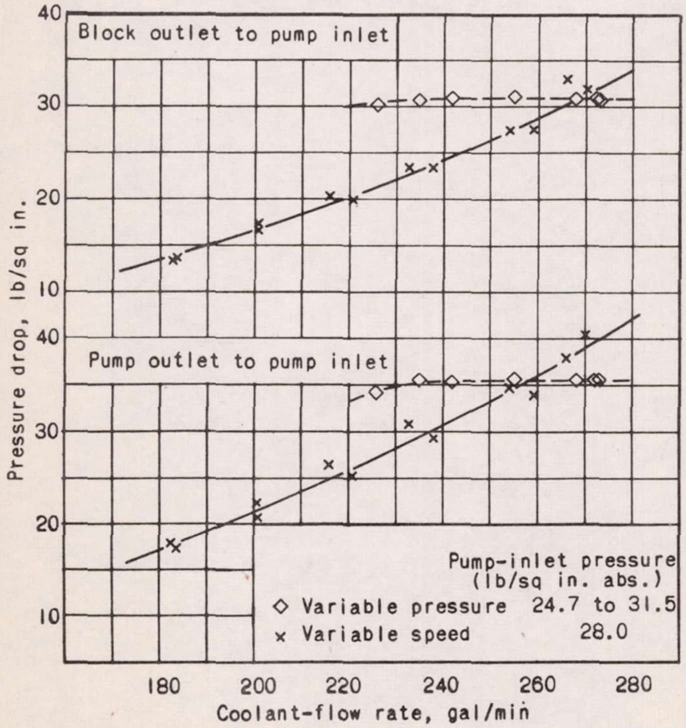


(b) Variation with pump-inlet pressure. Engine speed, 3000 rpm.

Figure 13. - Variation of coolant-flow rate through shunt lines. Coolant-out temperature, 250° F; coolant, 30 percent AN-E-2 ethylene glycol plus 70 percent water; cooling system B; pump speed, 1.234 times engine speed.

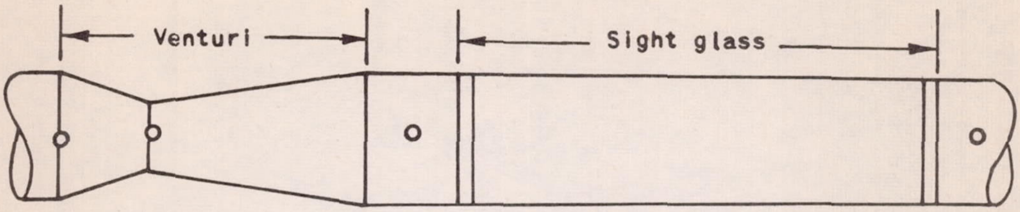


(a) Cooling system A.



(b) Cooling system B.

Figure 14. - Variation of pressure drop through entire system and from block outlet to pump inlet with total coolant-flow rate. Coolant-out temperature, 250° F; coolant, 30 percent AN-E-2 ethylene glycol plus 70 percent water.



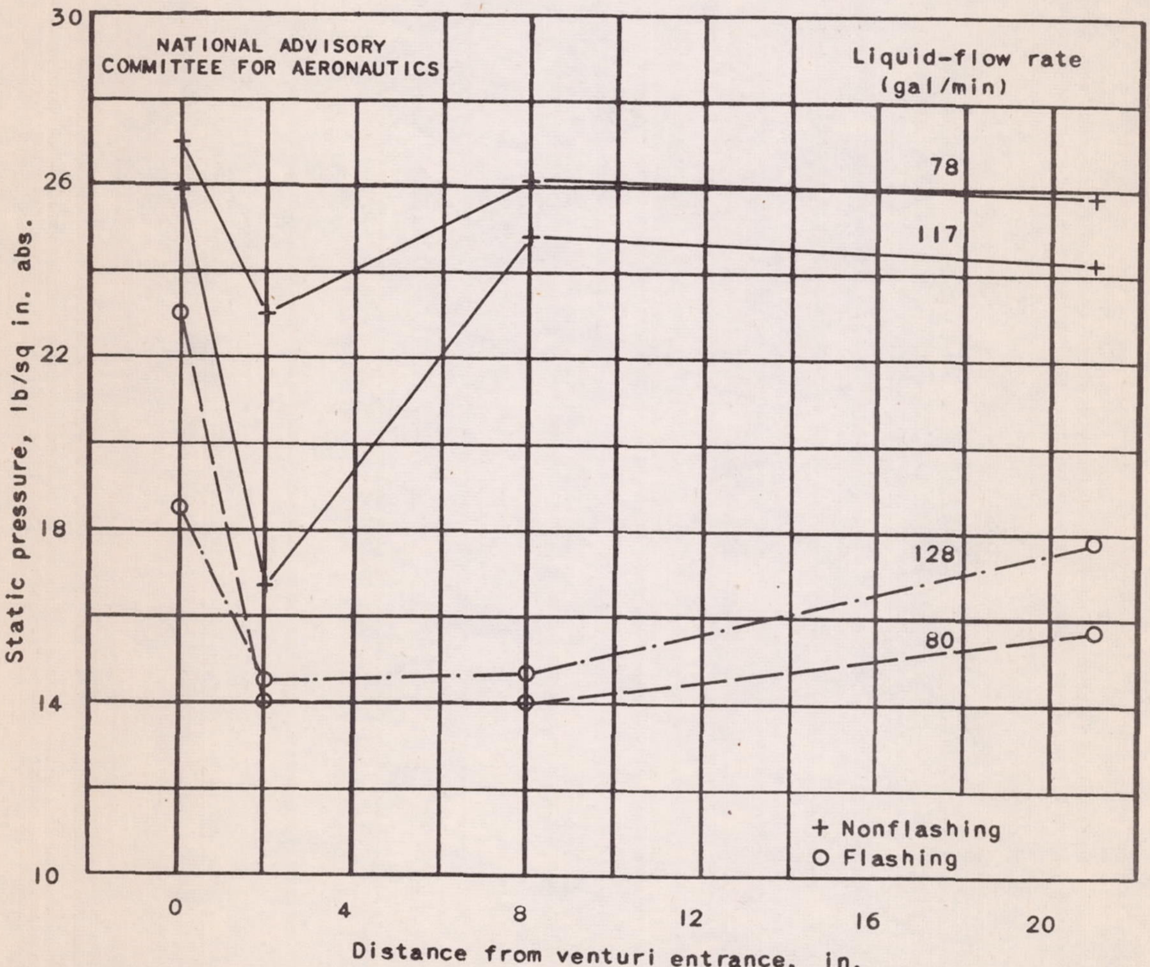


Figure 15. - Pressure survey through venturi and sight glass for flashing and nonflashing runs. Fluid, 30 percent AN-E-2 ethylene glycol plus 70 percent water; fluid temperature, 219° F.

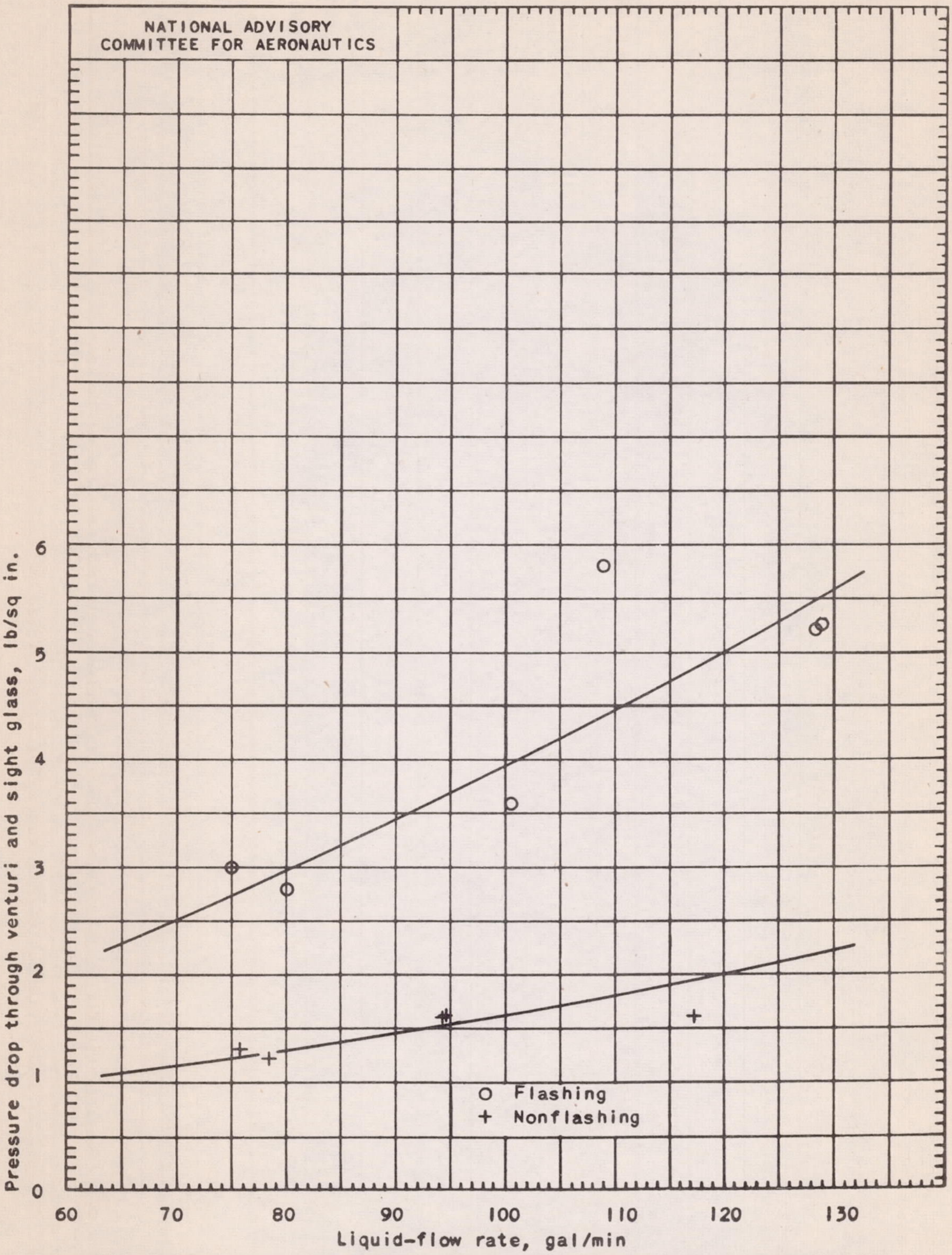


Figure 16. - Variation of over-all pressure drop through venturi and sight glass with fluid-flow rate. Fluid, 30 percent AN-E-2 ethylene glycol plus 70 percent water; fluid temperature, 219° F.

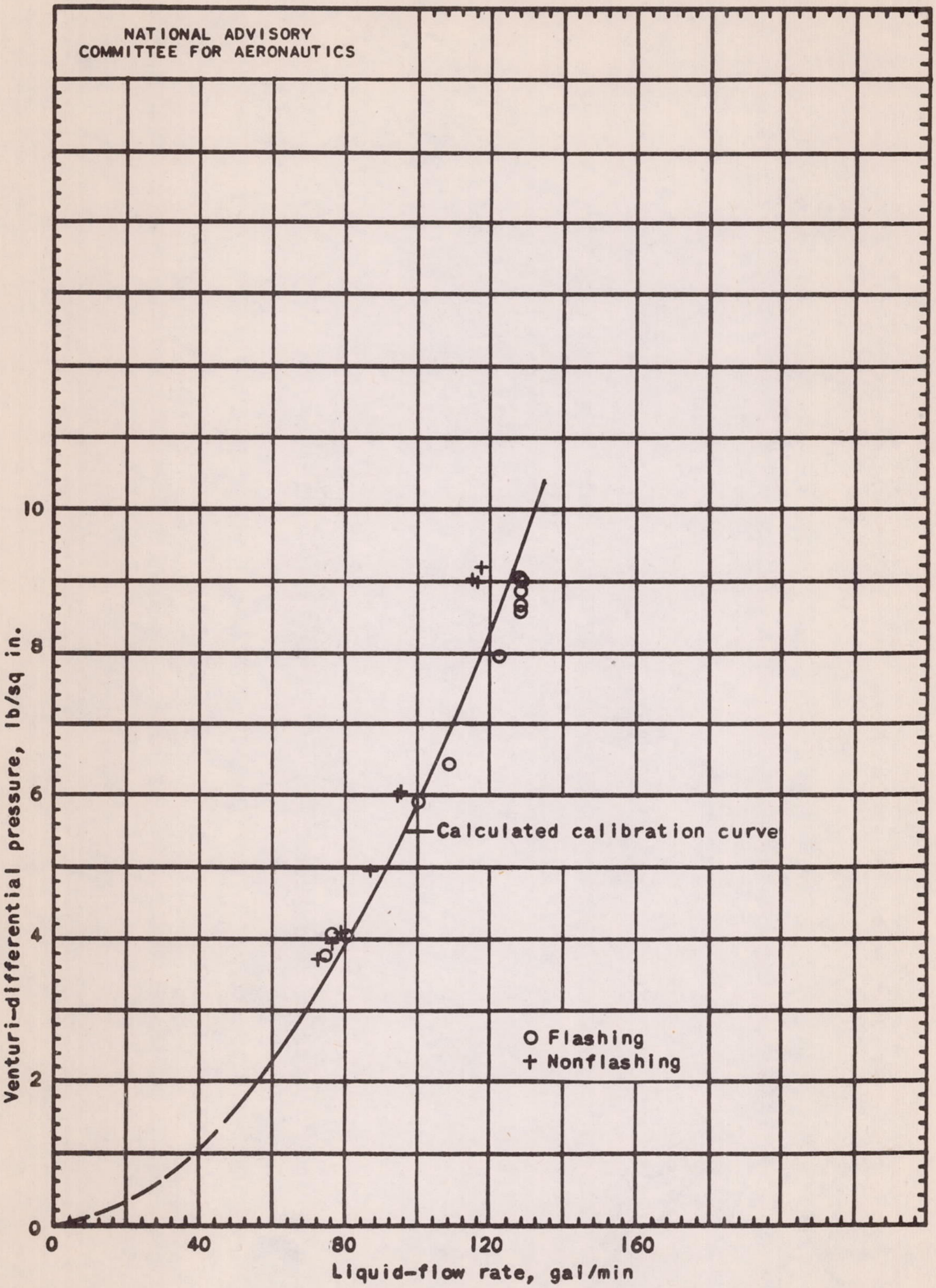


Figure 17. - Venturi calibration curve for flashing and nonflashing runs. Fluid, 30 percent AN-E-2 ethylene glycol plus 70 percent water; fluid temperature, 219° F.